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Adaptive flow optimization of a turbocharger compressor to improve engine low speed performance †

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Abstract

To improve the engine overall performance, an adaptive flow optimization procedure is proposed in this paper to synthesize turbocharger compressor optimum designs. Two objective functions are involved in the adaptive optimization. They are the traditional compressor design and the compressor design with consideration of improving engine overall performance. The two-step decomposition approach is chosen to generate optimum designs. The optimized designs not only satisfy turbomachinery and engine constraints but also have optimum objective function values in the two fields. Performance sensitivity analysis of compressor main design variables is performed for the flow optimization design process. A centrifugal compressor is redesigned for a turbocharged gasoline engine, as an example, based on the adaptive flow optimization process. The calculating results show a more than 5% increase of isentropic efficiency in comparison with the base line compressor, resulting in a more than 19% increase of engine torque at low speed conditions.

Keywords: Adaptive flow optimization; Turbocharger; Compressor; Low speed performance; Overall performance

1. Introduction

Turbocharging the internal combustion engine is as old as the engine itself. Early on, it was used to improve the high-altitude performance of aircraft engines and now is considered as a promising way for fuel energy saving and CO_2 reduction [1-6]. In spite of promising alternative developments of automobile applications, the turbocharged internal combustion engine will remain dominant for the foreseeable future, and the small aviation transportation development opens the door to a new era for the research in turbocharged internal combustion engines.

Turbocharger design is a major challenge for engine performance improvement [7, 8]. Designing turbocharger compressor and turbine blades is a complex task involving many different objectives and constraints coming from various disciplines. For example, the main performance design requirements of the compressor blades involve the required pressure ratio, higher thermodynamic efficiency, and adequate surge margin, etc. However, the higher efficiency will lead to decrease of the surge margin. As a result, the efficiency and surge-margin improvement must be compromised according to a certain design criterion. Much work has been reported on the application of optimization techniques to turbomachinery design [9-13].

During the traditional turbocharger blade design process, the turbocharger is designed and developed by turbomachinery experts. It is obvious that all these optimizations considered the objective and constraints in the field of turbomachinery only [14-17]. However, a turbocharged internal combustion engine is the combination of a reciprocating piston engine and rotating turbomachines. The operation of turbomachines is fundamentally different from that of reciprocating engines, so a turbocharger is mismatch to the engine when the engine runs away from the turbocharger design point, which leads to poor engine performance.

In order to improve the turbocharged engine overall performance, especially the performance at low speed conditions, the turbocharger design optimization including the consideration of engine operation requirements has to be employed. This paper is to create a procedure to use adaptive optimization technique to design the turbocharger compressors. The compressor and the engine operation requirements considered in the optimization are the compressor design point and the engine maximum torque operation point. Because the two points involved are well defined and can be separated easily, the two-step decomposition approach is applied to solve this adaptive optimization problem.

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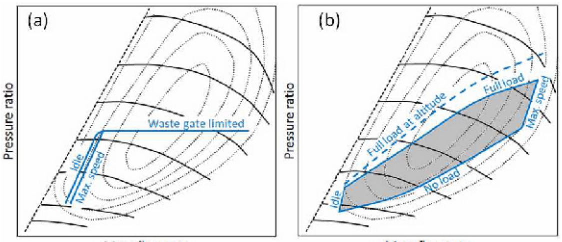


(a) Gasoline engine (b) Diesel engine

Fig. 1. Compressor operation with a turbocharged engine.

2. Adaptive flow optimization approach

The turbocharger compressor will operate over a wide range of speeds and mass flow rate conditions when equipped with an internal combustion engine. A turbocharger compressor operating with a gasoline engine or diesel engine is illustrated in Fig. 1 [18]. Usually the turbocharger design optimization is mainly focused on the engine-turbocharger matching operation point requirements. A rational turbocharger compressor design process should assist the designer in ensuring that the design fits the engine non-matching operation point requirements and brings the engine concerns into the compressor design process. It is obvious that the optimization to adaptive engine operation requirements has to be employed.

To increase engine low speed torque is a major challenge for a turbocharged engine design. The engine non-matching operation requirements considered in the compressor adaptive optimization are the requirements at the engine maximum torque point. This is corresponding to the compressor low flow rate point, which can be determined by the engine cycle simulations. Here this compressor operation point is named as point A, and the compressor design point, which related to engine-turbocharger matching operation point, is named as point B for simple.

The adaptive optimization is decomposed into two steps. A general two-step decomposition formulation for this paper is given as follows.

Step 1

Minimize $F_B(X^B)$. (1)

Subject to

$$g_i^B(X^B) \le 0; i=1,2,...,k \tag{2}$$

$$\sum_{i=1}^{n_B} \frac{\partial F_A}{\partial X_i^B} \Delta X_i^B \le \varepsilon_A \tag{3}$$

$$X_j^{Al} \le X_j^A + \sum_{i=1}^{n_B} \frac{\partial F_A}{\partial X_i^B} \Delta X_i^B \le X_j^{Au}; \quad j=1,2,...,n_A \tag{4}$$

$$X_i^{Bl} \le X_i^B \le X_i^{Bu}; i=1,2,...,n_B. \tag{5}$$

Step 2

Minimize $F_A(X^{B*}, X^A)$. (6)

Subject to

$$g_i^A(X^{B*}, X^A) \le 0; i=1,2,...,l \tag{7}$$

$$X_i^{Al} \le X_i^A \le X_i^{Au}; i=1,2,...,n_A \tag{8}$$

where F_B is the objective function of the first step, which is related to compressor performance of point B. X_B is the design variable associated with the performance of point B, such as impeller inlet blade angle, tip and hub radii, impeller outlet blade angle, outlet radius and width [20, 21]. g_i^B is the constraint for the turbocharger compressor design, such as surge and stall margins. F_A is the objective function of the low flow rate point (point A). ε_A is a given constant to limit the variation of F_A due to changes of X_B. X_j^A is the jth design variable associated with the performance of point B. X_j^{Al} and X_j^{Au} are the lower and the upper limits for X_j^A, respectively. X_i^B is the ith design variable associated with the performance of point B. X_i^{Bl} and X_i^{Bu} are the lower and the upper bounds for X_i^B, respectively. X^{B*} is the optimum design with the performance of point A. g_i^A is the constraint for point A optimization . l is the number of constraints in the point B optimization step.

The first step is mainly focused on the performance at point B, and the adequate margin of surge and choke are considered as the flow optimization constraints. This step is in unison with traditional compressor design optimization, and can be done in the field of turbomachinery itself. Eqs. (3) and (4) can be considered as move limit constraints. The optimization process begins with the first step. The optimum design variables obtained by the first step are passed to the second step.

To increase engine low speed torque is a major challenge for a turbocharged engine design. The second step optimization is mainly focused on the performance at point A; the pressure ration at point B, and the margin of surge and choke obtained by the first step are considered as the constraints of the second step flow optimization. Because the engine has enough exhaust energy to drive the compressor at the engine-turbocharger matching operation condition, the compressor efficiency obtained by the first step at point B can be eliminated from the constraints of the second step optimization. The compressor efficiency affects engine performance greatly at engine low speed operation points because of the lack engine exhaust energy. Therefore the second step optimization proceeds to maximize the compressor efficiency at point B for the engine performance improvement at engine low speed conditions.

3. Sensitivity analysis of design variables

Eq. (3) prevents the large change of F_A due to the large change of X_i^B, and this increases the chance of convergence in the second level optimization. Eq. (4) is used to ensure the second step design variables within their bounds when the design variables change in the first step. A compressor through flow model is used to investigate effects of the change of design variables on compressor performance [22, 23], and a design of experiments (DOE) method has been performed to study the weights and effects of different design data [24].

Table 1. Turbocharger compressor operation conditions.

Conditions	Rotational speed(RPM)	Mass flow(kg/s)
A	80000	0.04
B	120000	0.10
C	160000	0.16

Table 2. Design variable changes for sensitivity study.

Design variables	Low value	Base value	High value
D_1/D_2	0.681	0.758	0.835
d/D_2	0.217	0.255	0.294
$\beta_{1,Tip}$ (°)	-59	-69	-79
$\beta_{1,Hub}$ (°)	-30	-40	-50
b_2/D_2	0.101	0.140	0.178
β_2 (°)	-35	-45	-55

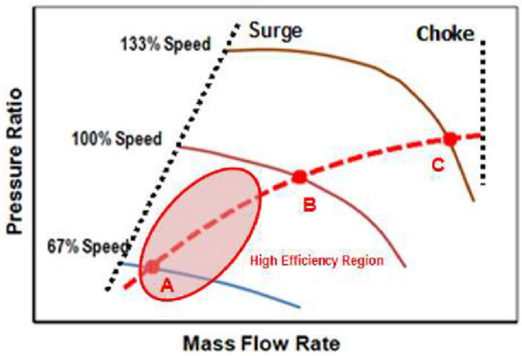
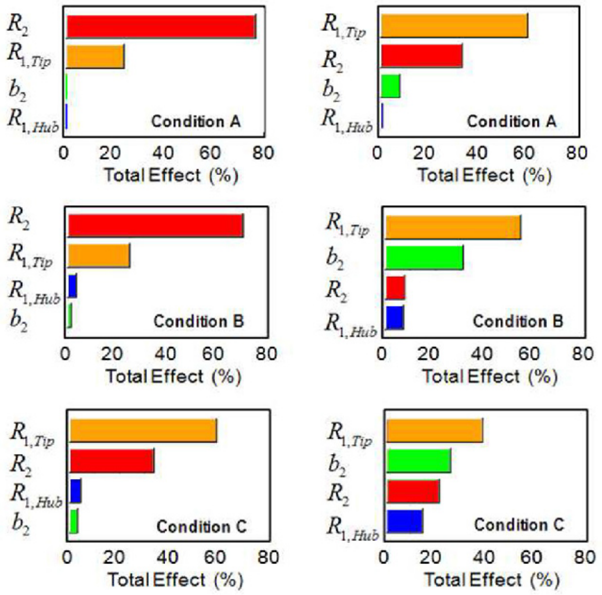


Fig. 2. Turbocharger compressor operation points sketch.

This is done with the aim of giving a justification about the choice of the design variables X_i^B and X_j^A.

The most important layout criterion for automotive propulsion is the achievable charge pressure in the speed range below maximum torque [25]. A gasoline engine turbocharger compressor is taken as a numerical example in this paper, and the point A and the point B conditions of the compressor operating with the engine are shown in Fig. 2 and illustrated in Table 1. The point C condition is the compressor operation point related to the engine maximum power point. The change of the compressor performance at point C might result in the change of compressor choke margin, and affect the engine maximum power output. Due to this reason, the point C is introduced for the sensitivity investigation. The rotating speeds at point A, point B and point C are 67%, 100% and 133% of the compressor design rotating speed, respectively.

The impeller inlet tip radius $R_{1,Tip}$ and angle $\beta_{1,Tip}$, the inlet hub radius $R_{1,Hub}$ and angle $\beta_{1,Hub}$, the impeller exit radius R_2, width b_2 and back sweep angle β_2, and the inlet mean section blade angle $\beta_{1,M}$ are investigated here. The originally design variable values of the gasoline engine turbocharger compressor are taken as the base values. The main design variable values are changed by about $\pm10\%$ for the high level (1) and low level (-1) separately for sensitivity study and Table 2 gives out the details.



(a) Pressure ratio (b) Efficiency

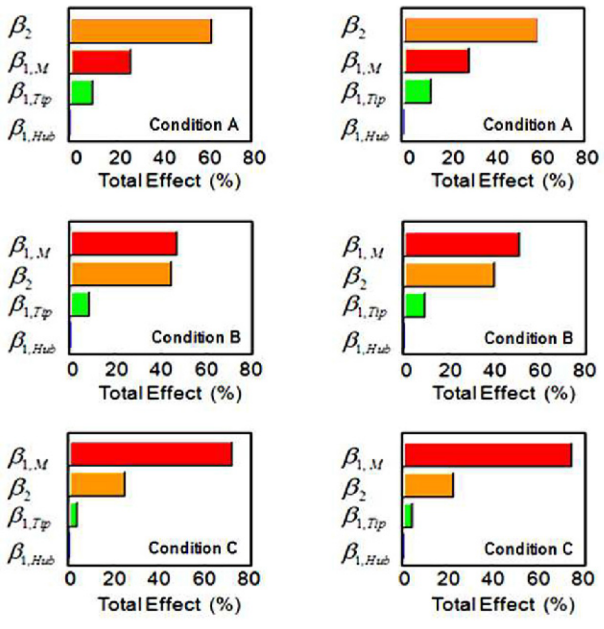
Fig. 3. Sensitivity investigation of the blade size variables.

The Pareto chart is used here to summarize graphically and display the relative importance of different variables. The horizontal axis of the Pareto chart is labeled as the effect percentage of particular variables on specific compressor performances (total effect).

Fig. 3 shows the sensitivity and effects of blade size design variables $R_{1,Tip}$, $R_{1,Hub}$, R_2 and b_2 on the compressor pressure ratio and efficiency performance at point conditions A, B, and C, respectively. It can be seen that R_2 affects pressure ratio greatly at points A and B, $R_{1,Tip}$ affects the pressure ratio greatly at point condition C, and $R_{1,hub}$ and b_2 have little effects on the pressure ratio. For the efficiency performance, $R_{1,Tip}$ has significant effect at all the conditions. $R_{1,Tip}$ has positive effects on the efficiency on its high level at condition B and C, but negative effect at condition A. The influence of b_2 on efficiency increases with the rotational speed and mass flow rate.

Fig. 4 gives out the sensitivity analysis of the blade angle variables $\beta_{1,Tip}$, $\beta_{1,Hub}$, β_2, and $\beta_{1,M}$ on the compressor pressure ratio and efficiency performance. The results show that $\beta_{1,M}$ as well as β_2 affect the pressure ratio and efficiency greatly. $\beta_{1,Hub}$, and $\beta_{2,Tip}$ show less influence on the general performance of the centrifugal compressor.

According to the sensitivity analysis, $R_{1,Tip}$, b_2 and $\beta_{1,M}$ are the main design variables X_i^B associated with the performance of point B, and will be determined in the first step optimization; $R_{1,Tip}$, R_2 and β_2, are the main design variables X_j^A associated with the performance of point A, and will be optimized in the second step. Although the two sets of design variables are optimized separately in the two steps, they are indirectly coupled through Eqs. (3) and (4).



(a) Pressure ratio (b) Efficiency

Fig. 4. Sensitivity investigation of the blade angles.

4. Gasoline engine compressor as an example

The gasoline engine turbocharger compressor, which design variables sensitivity has been analyzed above, is taken as a numerical example for the adaptive flow optimum design to improve engine performance at low speed conditions. This 1.8 liter engine is an in-line four-cylinder turbocharged, inter-cooled, four-stroke gasoline engine which operates from 880 to 6000 rpm.

4.1 Compressor adaptive optimization

As discussed above, the adaptive flow optimization is decomposed in two steps. For this compressor, the first step is in unison with traditional compressor design optimization, and has been done by the turbocharger manufacture itself. The optimum design compressor from the manufacture is taken as the prototype in this paper. The adaptive optimization can begin directly from the second step optimization process.

In the second step optimization is mainly focused on minimizing the flow loss at point A, i.e., to maximize the efficiency at point A. According to the sensitivity analysis, $R_{1,Tip}$ R_2 and β_2, are the main design variables X_j^A associated with the performance of point A, and have been optimized. Table 3 represents the results of the originally prototype design by the manufacture and the new design with the adaptive flow optimization in this paper. Fig. 5 shows the detailed impeller geometry of the adaptive optimization compared to the prototype design and Fig. 6 gives out the comparison of the impeller angle distributions.

Table 3. Compressor design variables.

Design variables	Prototype	New design
$R_{1,Tip}$ (mm)	19.7	18.5
R_2 (mm)	26	25
β_2	-45°	-30°

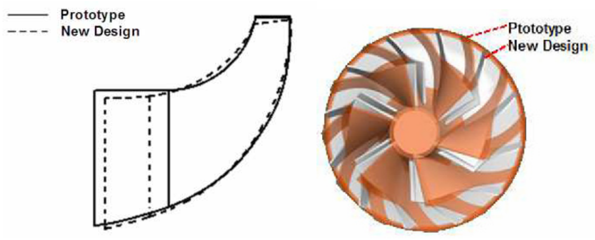


Fig. 5. Impeller geometry of prototype and new configurations.

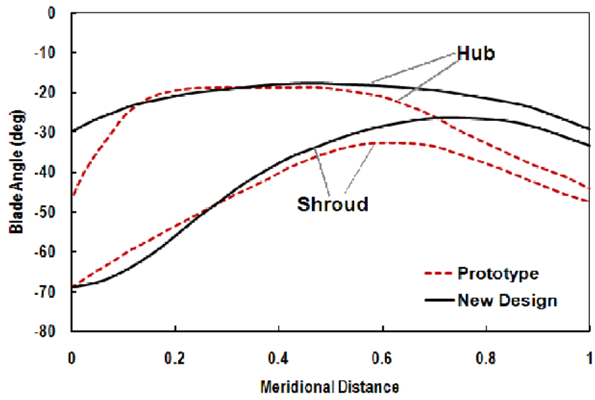


Fig. 6. Impeller angle distribution of prototype and new compressors.

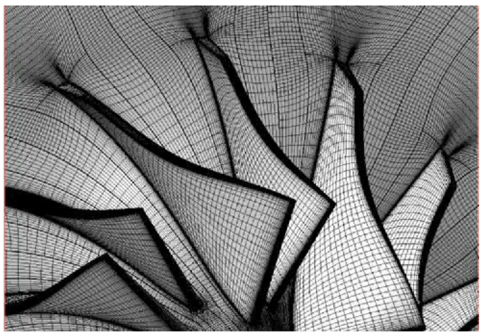


Fig. 7. 3D view of prototype compressor multi-block grid.

4.2 Flowfield and performance analysis

The compressor flowfield and performance of the prototype and the new design by the adaptive optimization have been investigated with CFD simulations using NUMECA code. To validate the CFD code, the multi-block grid of the whole compressor stage has been built as indicated in Fig. 7. The characteristics of the prototype impeller at 90000 rpm have been evaluated in details and are plotted in Fig. 8 compared

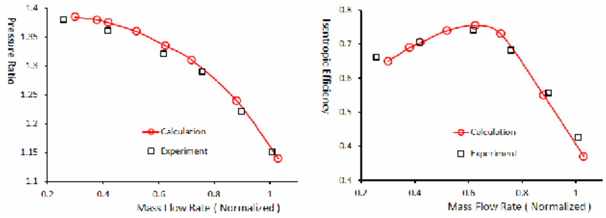
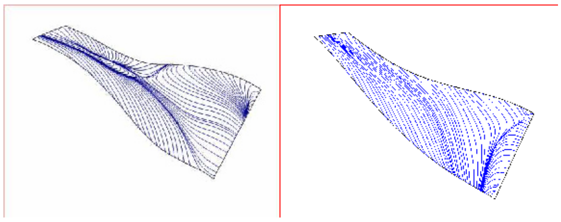


Fig. 8. Comparison between numerical and experimental results.



(a) Prototype (b) New design

Fig. 9. Near wall streamline on the blade suction side.

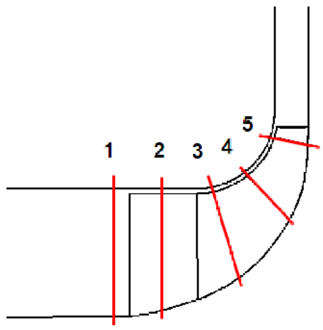
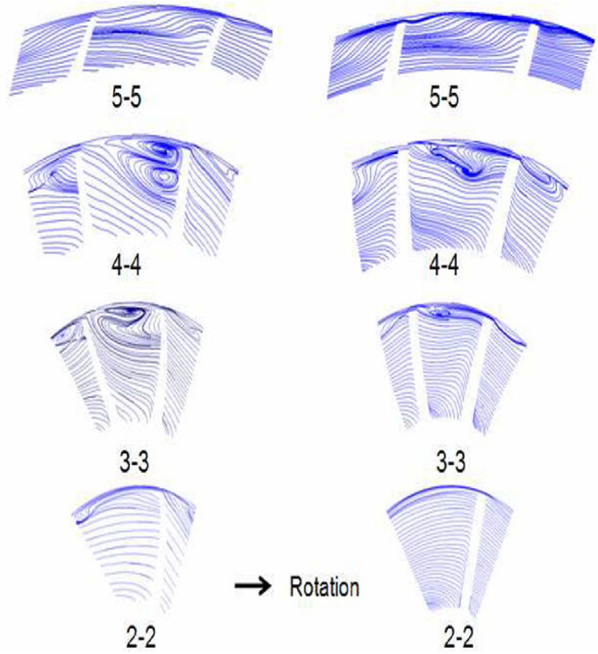


Fig. 10. Compressor passage cross sections.

with the experimental measurements. The pressure ratio prediction values show good agreement with the experimental data. For the isentropic efficiency, the concordance between the numerical results and the experimental measurements are within 5% at different mass flow rates. The computational model is credible for the performance prediction.

Fig. 9 presents the near wall streamlines on the suction surface at point A operating condition of 80000 rpm and 0.04 kg/s. A separation region starts from proximately 50% chord near the shroud of the prototype and indicates a strong vortex flow from the suction side. The separation is obviously restrained by the adaptive optimization for the new designed blade. Fig. 11 gives out the cross streamlines of the compressor passage cross sections illustrated in Fig. 10, as well as the blade rotates in the clockwise direction. It is noticed that in Fig. 11 a vortex flow is generated on prototype section 3-3 and developed on section 4-4, but the vortex flow behavior is weakened on new design compressor sections 3-3 and 4-4. This is to increase the compressor efficiency at point A. Fig. 12 shows the efficiency of the new designed impeller at point A operation condition increases more than 5% from its initial value of prototype. While changing the flow rate along the



(a) Prototype (b) New design

Fig. 11. Secondary streamlines at point A condition.

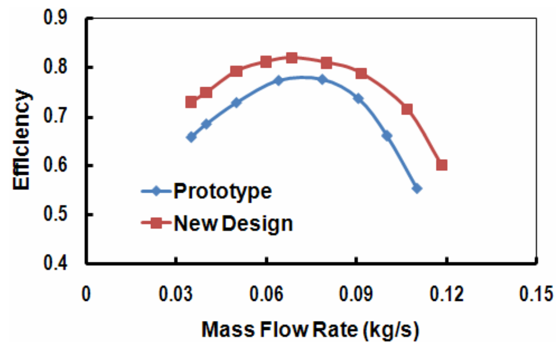


Fig. 12. Compressor efficiency with constant rotating speed.

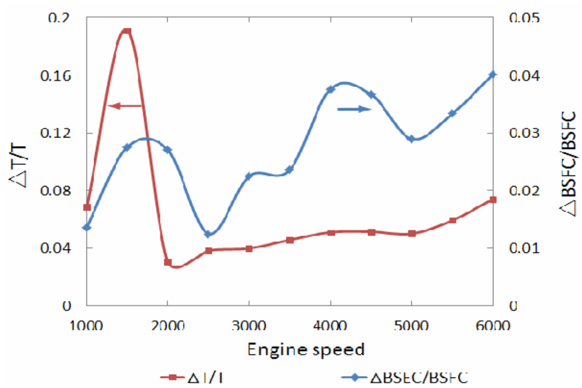


Fig. 13. Predicted engine performance comparison.

point A speed line of 80000 rpm, the efficiency increases significantly.

Fig. 13 gives out the predicted engine performance comparison between the prototype and the new design of turbo-

charger compressors matching with the engine. The torque of the engine at point A operation condition has been increased by about 19% by the new adaptive optimization design. The notable reduction of the brake specific fuel consumption is also seen at the medium and high engine speeds with the adaptive optimization, for example, by about 4% at point B operation condition.

5. Conclusions

Turbo machines and internal combustion engines are two well-developed research areas. The objective of optimization and constraints in two areas are usually different. The optimization of a turbocharger design is usually down in the turbomachinery field itself. As a matter of fact a turbocharger is also an engine component which can change the engine operation from time to time. Considering this nature of turbocharging a new idea of turbocharger compressor adaptive flow optimization with consideration of engine operation requirements is proposed in this paper. The two-step decomposition approach is chosen to generate optimum designs. The optimized designs not only satisfy turbomachinery and engine constraints but also have optimum objective function values in the two fields. To deal with difficulty of optimization with design variables, sensitivity analysis of the main design variables is performed. A gasoline engine turbocharger compressor is taken as an optimization example, and the numerical results show that the approach is effective. It is believed that the suggested new approach can be used effectively to improve the overall performance of turbocharged internal combustion engines, especially the engine low speed performance.

Acknowledgments

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